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TITLE: EFFECT OF TIRE DYNAMIC PROPERTIES
ON AIRPLANE LANDING GEAR SHIMMY

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PREPARED BY

N. C. McNeil, Jr., H. J. Donald

SUPERVISED BY

H. J. Donald

APPROVED BY

H. J. Donald

APPROVED BY

H. J. Donald

H. J. Donald

10-9-70

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FOREWORD

The following paper was prepared by H. C. Attari and H. C. [unclear] for presentation at the EAS Committee (Aerospace Landing Gear Systems) meeting. The paper was presented on October 14, 1972.

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REPORT ON THE FLUIDIC PROPERTIES
ON AIR FLOW IN A GEAR THROAT

H. C. Gentry
Senior Engineer

H. Howard
Senior Engineer

The Boeing Company
Seattle, Washington

ABSTRACT

The self-excited vibration of a landing gear (shimmy) is generated in an interplay of effects present in the tire, the landing gear proper, and in the entire airframe itself. This paper reviews the significance of the various parameters and urges an industry wide effort to enable better accounting of the tire influences. 7/4

SUMMARY

Airplane Landing Gear

Shimmy is the unwanted multi-degree of freedom oscillation of the landing gear arising mainly as a result of ground excitation forces. Such vibrations can sometimes assume grave proportions. Shimmy has been investigated by many researchers in the past twenty years and this problem still defies a satisfactory solution. This paper reviews some of the prevalent views on the subject. The major short-coming of the technical approach to this problem is the lack of adequate understanding of the dynamic properties of tires. In the absence of it, the tire models have been an evolutionary hodgepodge. This is illustrated by a review of the tire models used by investigators. To produce a positive means of resolving this problem, which has plagued a large number of aircraft, an experimental program is suggested. This will help determine the dynamic properties of tires and develop transfer functions to enable credible analysis of the response of landing gear to forces and moments arising out of single or multiple wheel assemblies.

DEFINITION

Self-excited vibration in a landing gear (shimmy) is initiated and sustained by the ground excitation forces. The problem is further complicated by increased emphasis on reducing the weight of the landing gear system in modern aircraft and the effects of this emphasis on gear structure. Because of its frequent occurrence and the seriousness (cost, safety, etc.) of the problem, analysis for the prediction of shimmy or landing gear instability is an important part of landing gear design. It also serves as a guide to shimmy testing programs. However, often only limited success is achieved in shimmy analysis and because of over-simplification of many variables as well as lack of meaningful data on tires. At least two forms of shimmy have been discovered (1, 2). These are:

- o tire yaw shimmy
- o structural torsional shimmy

The occurrence of both forms depends on the amount of overall systems damping. For low damping, shimmy usually involves rigid-body torsional motion of the landing gear restrained by the pivoting motion of the tire. That is, the nodal mass is essentially pivoting about the swivel axis, and the effective spring rate is the tire dynamic-torsional spring rate, accounting for the radial distance of the footprint lateral load line of action from the swivel axis. This type of shimmy is referred to as TIRE YAW SHIMMY.

With high damping, tire yaw shimmy is stabilized but the structural modes of the gear can become unstable. The gear's occurring mode has essentially the same nodal mass as the first mode; however, the spring rate is the torsional-spring rate of the

*Numbers in parentheses refer to references in the attached reference list.

landing gear with the damper (if any) locked. This type of shimmy is often referred to as structural-torsion shimmy. Both types of shimmy are affected significantly by the tire parameters.

If a precise mathematical description of the shimmy phenomenon were attempted, it would involve so many variables that solution of the problem and interpretation of the results would become impractical if not impossible. However, the difficulties in the solution are due merely to the number of factors, and, if it can be established that some of these are more significant than others, the problem can be simplified. In recent years efforts have been made to accomplish this. The valid simplification in modeling the landing gear components can be made if concerted effort is made to obtain meaningful data on several significant parameters.

Any theory which provides a mathematical description of shimmy motion has to date almost always been restricted to linear cases, e.g. see appendix II. Nonlinear elements, such as tire characteristics, friction, and clearances, however, are essential in determining shimmy behavior of landing gears. It is the introduction of these nonlinearities, however, which increases the complexity of the problem considerably.

To represent certain sub-components by more realistic functions more accurate information about these components is absolutely necessary. The accuracy of the prediction of shimmy by analytical means will increase with more accurate correspondence between the mathematical model and the real mechanical system. A certain degree of approximation is necessary but the essential features of a real system have to be present in the model.

In the survey that follows, the various parameters that influence the shimmy phenomenon are considered in light of the above statements and special emphasis is placed on the dynamic properties of tires.

UPPER LIMITS OF SHIMMY

Analysis of shimmy must consider the following:

- o the nature of airframe flexibility
- o the nature of the attachment of the gear to the airframe
- o flexibilities of the attachment and gear structure
- o the interaction or coupling between the airframe and the gear structure
- o the nature of excitation forces and moments
- o the presence of freedom at all support points and joints
- o the interaction between the structure and the damping device

With these guide lines, the status of presently used shimmy analysis methods (shimmy theories) are reviewed in the following sections.

Airframe Considerations

It has been generally recognized that in order to understand the true nature of the self-excited vibration and its possible multiple modes of motion, the airframe inertia and elasticity must be considered. For example, Moreland (3,4) suggests that a direct method of accomplishing this is to set up the dynamic equations for the overall system and obtain the single higher order differential equation from which the stability criterion and frequencies may be obtained. Other investigators also

entertained this idea. For example, Pacejka (5) considers the inertia and elasticity of the frame in his analysis. Reference 3 contains several simpler models to enable consideration of inertia and flexibility.

The general equation can be expressed in the following form:

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = \{F\}$$

where $\{F\}$ is largely determined by the excitations transmitted by the gear structure.

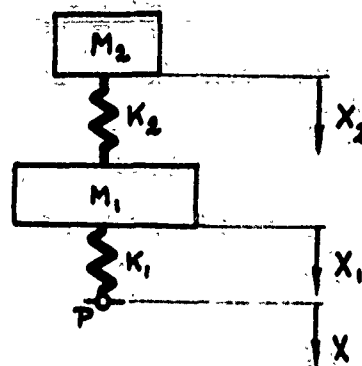
The simpler versions consider the airframe merely as a rigid mass. A slight modification of this is a version which considers the airframe in terms of an equivalent lumped parameter system as shown in the figure below. The joint P represents the attachment point of the equivalent elastic element and the oleo. K_1 represents the combined elasticity of the oleo and the torsional elasticity of the fuselage. M_2 and M_1 are lumped airframe masses. The equation of motion can be expressed as follows:

Considering the forces on M_1

$$M_1 \ddot{X}_1 = K_1(X - X_1) - K_2(X_1 - X_2)$$

and for M_2

$$M_2 \ddot{X}_2 = K_2(X_1 - X_2)$$



Similar three and four degree of freedom models are discussed by Moreland (3). In general, this aspect is treated quite well by the airframe structural dynamicists.

Attachment Considerations

The hinges and universal joints are generally represented as developing no rotational stiffness in desired directions for beams entering these joints. The attachment frame is generally simplified by replacing complex shape members with flat plates or circular pipes. The inner cylinder is generally idealized as a beam, with the bearing plate connection consisting of a number of stringer elements (axial load only) per bearing plate. This provides the required bearing area, as well as preventing swivel moment transfer from the inner to the outer cylinder through anything other than the torque links. The landing gear structure is sometimes extended to include the landing gear support beam which is sometimes pinned at both ends. Other support points may include the rear spar end of the trunion (free to rotate about the trunion centerline), the side strut apex (free to rotate in the plane of the side strut), and the actuator support (completely fixed). Implications are also essential due to the presence of a steering system (on nose as well as steered main gears). The steering cylinders are generally idealized as stringers.

Boeing has used, with considerable success, the concept of influence coefficient in evaluating structural influences. The Boeing tests consisted of mounting the gear and support beam in a jig, applying loads and measuring deflections. The data was plotted and reduced to influence coefficients. Sufficient loading conditions were tested to provide data crosschecks, and these indicate that the data is quite accurate. The data obtained from plastic models provides the best comparison to the data.

obtained for this test.

In addition to its use in the shimmy analysis, these data can also provide a basis for developing a mathematical model to determine the influence coefficients of other landing gears. Moreland was the first to suggest the use of the transfer function approach to evaluation of the structural influences, although he never included it in any of his analyses. This approach still merits consideration.

Presence of Freeplay

Due to the presence of a large number of moving and restrained support points and a rather severe impact and vibration environment, the development of freeplay at some support points, such as the trunnion, is inevitable.

This unavoidable freeplay has a significant effect upon the magnitude of the perturbations which an otherwise stable landing gear can tolerate. Landing Gear shimmy simulation with a prescribed freeplay poses a formidable problem. Reference 6 attempts a reasonable approximation. In this analysis, mechanism joints with specified freeplay in the side, fore and aft, and rotational directions were considered.

Trail Stabilizing or Destabilizing

Several investigators have considered the contribution of mechanical trail to landing gear shimmy. The derivative of the critical damping ratio with respect to the trail generally can be either positive or negative. It thus follows that increasing the trail from zero may require either more or less damping which will depend upon the relative magnitudes of the influencing variables. For example, Moreland showed:

$$\frac{\partial R}{\partial L} = \frac{C_c V K_1}{B K_t^2} - \frac{K_t}{2 C_c V}$$

B = Ratio of $\frac{C_t}{C_c}$ - damping ratio

L = Trail of wheel axis behind axis of rotation of swivel

C_c = Reference damping coefficient

C_t = Torsional damping coefficient

J_w = Diametrical moment of inertia of wheel

k_1 = lumped airframe and gear elasticities

k_t = torsional spring constant of the gear.

V = Forward velocity of airplane.

Moreland (3) in his studies showed plots of R as a function of trail ratio

($T = L \sqrt{\frac{k_t}{k_s}}$). He observed that for values of inertia ratios less than unity, stability is possible for all values of trail - the required damping usually increasing with trail to a maximum and thereafter approaching zero asymptotically. For values of inertia ratio slightly above unity the range of damping for stability is limited.

SHIMMY DAMPER

A shimmy damper is often an agency requirement for averting the occurrence of shimmy. Damping as such, however, is not a certain means for avoiding instability. The system stability or limit cycle can be assured, however, by use of an appropriate type and amount of damping. Analysis of the 737 main landing gear showed that both excess and deficit damping would result in an unstable gear. The choice of a v^2 damper is not recommended for all configurations.

Analyzing the stability of a typical landing gear system having a damper in series with the torsional elasticity of the strut shows that both types of shimmy can occur. For low damping, tire yaw shimmy can occur with the dynamic torsional and lateral spring characteristics of the tire or tire cluster as the effective elasticity.

For high damping, the structural modes of the gear can become unstable. Usually the torsional spring rate of the gear structure is the governing factor. Although a designer seeks clear-cut answers to the question of gear stability, these are not possible due to the highly non-linear nature of this problem. The stability criteria are true for the equations used to describe the system. They are, however, not necessarily valid for the actual physical system if too crude estimations were used to establish the analytical model.

In addition the Hurwitz criterion (4, 5, 11) is used to calculate the stability limits. This criterion is relatively easy to handle mathematically. It provides an accurate stability limit but does not provide any means of quantitative stability assessment. Other more powerful methods of predicting stability have not found favor with shimmy researchers. This is understandable because historically landing gear has been treated as a structure rather than a system.

TIRE

The importance of the tire to the system and the shimmy phenomenon is not equally appreciated by various investigators. In recent years, however, the importance of the tire in this self-excitation phenomenon has been more and more recognized (1, 5, 6). The difficulty in incorporating an appropriate tire model in the shimmy analysis lies in the lack of knowledge about the dynamic behavior response of a rolling tire to an oscillatory side and angular motion. In most theories the tire is represented as a linear a torsional and/or lateral spring-damper system for which static or low speed rolling tire data are used.

Table I is a summary of the tire models used in various shimmy theories. The table indicates that the analytical models of the tire used for these analyses are

largely based on parameters which can be measured easily on a stationary (non-rolling) tire. Some low speed rolling data are also added. These parameters and the associated mathematical relationships are, however, undoubtedly only gross approximations of the fundamental mechanics for rolling pneumatic tires.

The table shows that the improvements made to the tire equations in course of time are more conjecture than real representation of the tire dynamics. In some cases terms are added merely to self cancel or influences are added twice inadvertently. Several of the tire parameters used are not independent of each other and depend on a variety of external conditions such as vertical load and rolling velocity (7, 8). Even so-called "dynamic" tire parameters obtained from vibration tests on stationary tires are considerably different from those obtained for the rolling tire (9, 10). It is, therefore, obvious that the approach thus far for the representation of the tire has resulted in inadequate representation of the exciting forces which arise from the tire contact area.

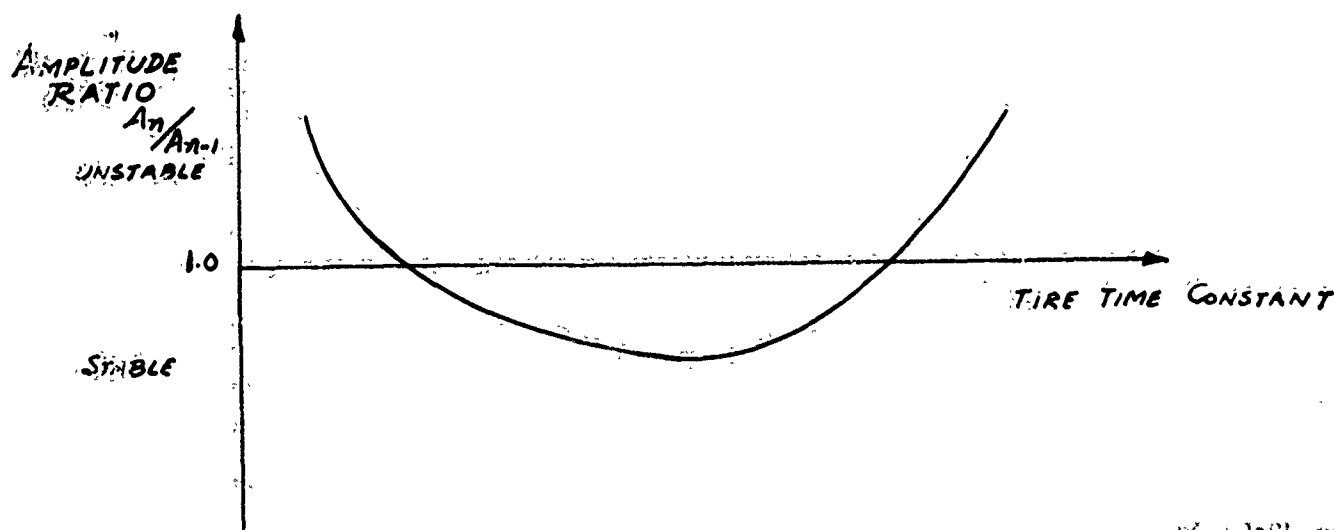
Considering these linear spring damper models for tire representation, the following paragraphs outline the significance of certain tire parameters upon gear stability. It can be readily comprehended, however, that these conclusions are only valid for the model used. If any one of the parameters is not constant but depends on other variables, this fact would have to be taken into consideration and modified stability criteria will apply.

Lateral and Yaw Stiffness of the Tire

Lateral and yaw characteristics significantly affect shimmy frequency and stability of the gear for tire yaw shimmy. Generally, increased lateral tire stiffness makes the gear more stable for this type of shimmy. Analysis indicates that increasing lateral stiffness is equivalent to increasing the dimensionless damping of the system and reducing its frequency. In the case of structural torsion shimmy, the frequency is not significantly affected by tire parameters. The stability boundaries, however, are greatly influenced by the tire stiffness.

Time Constant of Tire

The time constant of the tire also has a major influence on the tire yaw shimmy mode. Increasing its value usually has a stabilizing effect. But it is the combination of lateral stiffness and time constant of the tire which really defines the stability.



Pneumatic Trail

The pneumatic trail of a tire is another factor influencing stability as well as frequency. No generalized statement can be made about the influence of this parameter on shimmy. Only the combination of other tire characteristics and pneumatic trail will define the stability criteria and shimmy frequency. Changes in the pneumatic trail characteristics of the tire can change the stability in any direction depending upon other parameters.

To our knowledge, no attempt has been made to correlate this analytical model with actual tire behavior at high rolling speeds, controlling and measuring all significant variables. The correlation of shimmy tests with analytical results using this tire model can not give any valid conclusions because too many other parameters, which also have been highly linearized or whose values are uncertain, are masking the results.

Facts like these offer further proof that the tire and its characteristics have a significant effect on stability as well as shimmy frequency for both types of shimmy (1). Only an accurate representation of the tire will allow a reliable prediction of shimmy stability of a landing gear system.

To improve the accuracy of the analytical tire representation used in shimmy analysis, two approaches seek to be feasible:

- a. Full scale well-controlled tire experiments and the derivation of "transfer functions" for tire motion and forces. This approach treats the tire as a "black box."
- b. Development of a tire model based on a thorough analysis of the tire treating it as a three-dimensional structure of some kind. The finite length and width of the contact area as well as the stiffness of the various tire components would have to be taken into consideration.

The first approach will provide a tire model immediately applicable to landing gear shimmy analysis if tests are conducted properly. A series of tests will be required for each tire size.

The second approach could be based on the data available from the first one. After having established a valid tire model, testing can be considerably reduced and might be restricted to measuring significant tire parameters on a stationary tire.

CONCLUSIONS

Landing Gear shimmy is a complex nonlinear self-excitation phenomenon. From the discussion above the following can be concluded:

1. The various interfaces of the landing gear are reasonably well understood except for the dynamic properties of tires.

2. The results of simplified analysis are only valid for the model considered. The credibility of such analysis can only be improved by developing models that more closely describe real system behavior.

RECOMMENDATIONS

With these facts in mind, the following steps should be undertaken to improve the credibility of shimmy analysis results:

1. A transfer function approach should be adopted to better account for structural as well as tire influences.
2. A major, industry-wide program should be undertaken to determine the dynamic behavior of tires. Such a test program is outlined in the attached appendix I.

TABLE I

| I BOURCIER & CARBON | II MORLAND | III CHILLEY | IV PAGEKA | V COLLINS & BLACK | VI ROTTER |
|---|--|---|---|---|--|
| $K_{\lambda} \lambda_2 = F_{\lambda G}$ | $K_{\lambda} \lambda_2 = F_{\lambda G}$ | $K_{\lambda} \lambda_2 + C_1 \dot{\lambda}_2 = F_{\lambda G}$ | NONE | $K_{\lambda} \lambda_2 + C_1 \dot{\lambda}_2 = F_{\lambda G}$ | $K_{\lambda} \lambda_2 + C_1 \dot{\lambda}_2 = F_{\lambda G}$ |
| $K_{\alpha} \alpha_1 = M_{\theta G}$ | (NONE) | $K_{\alpha} \alpha_1 + C_2 \dot{\alpha}_1 = M_{\theta G}$ | $N_{\alpha} \alpha_2 = M_{\theta G}$ | $K_{\alpha} \alpha_1 = M_{\theta G}$ | $K_{\alpha} \alpha_1 + C_2 \dot{\alpha}_1 = M_{\theta G}$ |
| $N_{\alpha} \alpha_3 = F_{\lambda G}$ | $N_{\alpha} \alpha_3 + N C_3 \dot{\alpha}_3 = F_{\lambda G}$ | (NONE) | $N_{\alpha} \alpha_3 = F_{\lambda G}$ | $N_{\alpha} \alpha_3 + N C_3 \dot{\alpha}_3 = F_{\lambda G}$ | $N_{\alpha} \alpha_3 + N C_3 \dot{\alpha}_3 = F_{\lambda G}$ |
| $\frac{\ddot{\lambda}}{C_{\lambda} V^2} = M_{\theta G}$ | (NONE) | $M_{\theta G}$ due to curvilinear rolling is included but influences tire path only through the linear relationship above. | $C_5 \frac{\dot{\theta}}{V} = M_{\theta G}$ | (NONE) | $\frac{\ddot{\lambda}_{\theta M}}{C_{\lambda} V^2} = M_{\theta G}$ |
| Plus geometric relationship: $\theta = \alpha_1 + \theta_{\theta M}$ | Plus: A non-slip equation equating angular rotation of the wheel about its pivot point to lateral velocity of the wheel divided by the trail arm. | Plus a kinematic equation correlating tire deflections with wheel coordinates through an assumed mathematical expression for tire path over the ground. | Plus a non-slip differential equation for the slip-lar motion of the wheel about its pivot point considering elastoh support of the latter. | Plus: A differential equation constraining the footprint center contact point to roll without slipping along the tire track. | Plus Kinematic Equations: $\lambda_3 = \lambda_2 + (\lambda_{\theta F} + \lambda_{\theta M}) - \lambda_{\theta G}$ $\theta = \alpha_1 + \theta_{\theta M}$ |

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APPENDIX I

DISCUSSION OF THE PROPOSED NEW DYNAMIC TIRE TEST PLAN: TIRE TRANSFER FUNCTION CONCEPT

In analyzing Landing Gear dynamic problems such as Shimmy or Ground Handling, the forces as well as moments originating in the tire footprint are transferred to the gear-truck pivot. Unless this transfer of ground influences is accounted for properly, the landing gear response cannot be predicted accurately. Present day data enables the tire to be represented as a linearized spring-damper model. There is no data available to establish the coupling terms which account for force or response interaction.

In order to establish interaction or coupling influence of changes taking place in the tread, it would be desirable to develop a transfer function approach to this problem. In other words, by varying parameters in the tire footprint (C) (See figure 1) it must be possible to assess whatever influences are transmitted to B in terms of triaxial forces, moments, and motion. On the other hand, variations of inputs at the axle center B will cause responses at the footprint C. This approach of developing transfer functions of the tire during the entire domain of operations would provide a satisfactory basis for future analysis. This concept has never been utilized for tire dynamics although in control dynamics it is used routinely.

The application of this concept for brake dynamics has provided a valuable tool for optimizing brake system performance. The use of this concept for tires and shimmy will no doubt provide a good insight into the nature of the nonlinearities and make the end result (data) more readily applicable to complex Landing Gear System problems.

DYNAMIC TIRE TESTS - GENERAL APPROACH

To establish a suitable two-way transfer function, all the tire variables must be measured in the footprint (C) as well as at the axle support (B). The variables measured at the axle support in B of figure 1 would include three force components, three moments around the three axes as well as the axle displacements in six degrees of freedom. The instrumentation at the axle support will minimize inertial influences. It is recognized that measurements in the footprint are more difficult but some assessment is necessary to establish tire dynamic response. The footprint deformations can be measured by using imbedded transducers and high speed photography. It is much easier to apply known inputs to the footprint and assess the results at (B) i.e. the axle center. The tire responses are highly speed dependent and thus all tests should be run for a range of preselected forward velocities.

In order to assess the coupling effects properly, the variables should be varied only one at a time and in a tire dependent manner. For example, the varying parameter could be varied as a sinusoidal, step or a ramp signal. At the outset only one tire size need be considered. The variables such as tread design, tread stock, surface finish and tire interactions should await later study. The test outline that follows discusses mainly three major tire dynamic responses, firstly independent of each other. These are:

1. Vertical load transmission
2. Braking response
3. Cornering

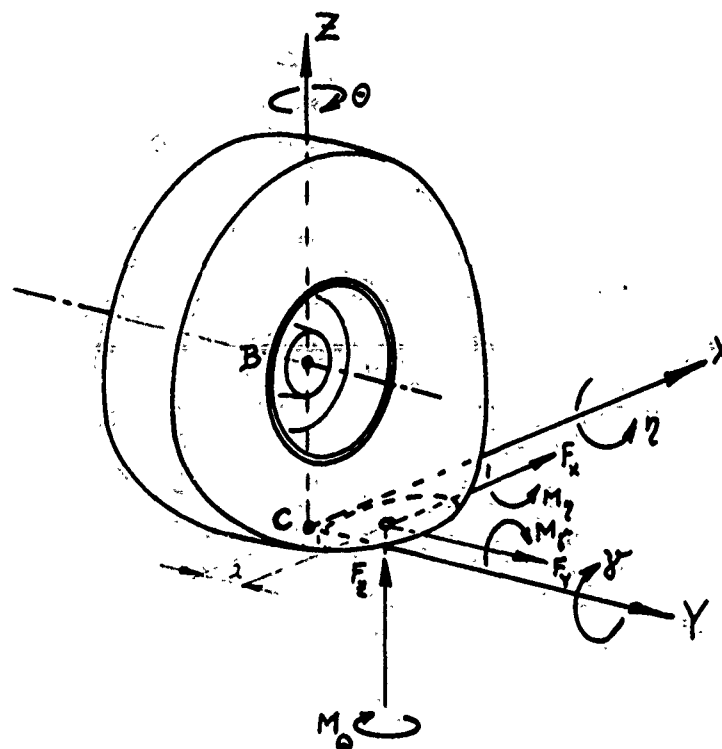


Figure 1

Once individual responses are established, it will be desirable to investigate the various coupling effects. This last aspect is very important in order to establish the validity of superposition or lack thereof. All these three dynamic tests should be conducted on both dynamometers and the NASA loads track.

PROPOSED TEST OUTLINE FOR DYNAMIC TIRE TESTS

1. Vertical Load Transmission

The dynamic response of an aircraft tire to changing vertical loads is considered to be the easiest dynamic property to investigate. Vertical load changes and deflection variations should be considered separately as inputs.

Vertical spring rate, wheel speed (rolling radius) and contact area should be evaluated.

It should be explored whether moving the tire center up and down while rolling on a flat surface will result in the same tire response as when the tire is excited by a certain pavement profile with the tire center held at constant height.

The influence of average load and value of deflection amplitude should also be investigated in the range up to 125% rated deflection. In addition, the validity of the superposition principles or lack thereof should be checked.

2. Tractive Force Response

As the next step, the fore and aft deflection characteristics of an airplane tire should be investigated. Cyclic brake torque would be applied as input. The maximum torque, however, should never introduce any noticeable sliding in the contact area.

The tire response would be evaluated by recording wheel speed changes, tractive forces, torque radius and footprint motion.

Again the influence of average torque and torque amplitude should be studied and the validity of the superposition principle checked for sinusoidal inputs.

3. Cornering Response

As this mode of tire response is highly critical for assessing the stability criteria of landing gears, a completely new approach should be taken toward this type of tire response. Several variables should be considered as inputs: yaw angle, steering moment, lateral motion, and side force. Again, input variations should be kept within limits that do not introduce any measurable sliding in the contact area. All forces, moments and motions should be recorded as outputs as well as the footprint displacements. For sinusoidal inputs the whole frequency range should be covered at certain forward velocities and the superposition principle checked. The influence of average values and amplitudes of the various inputs on the tire response should be evaluated. In addition the tread conditions should be carefully controlled.


STATIC TESTS

Static tire data also needs to be updated as it is widely used by Landing Gear Designers. It will also serve as a useful baseline while establishing correlation between dynamic (flat surface as well as road wheel data) and static data. The following variables should be measured.

1. Vertical spring rate
2. Lateral spring rate
3. Fore and aft spring rate
4. Torsional spring rate
5. Decay length
6. Cornering power (for low speed rolling)

APPENDIX II

It is not the intention of this paper to derive the differential equations necessary to describe the whole landing gear system for study of its instability criteria. This can be found extensively in the pertinent literature (4, 11).

For the sake of completeness, however, a full set of equations in Matrix form is given in Table II. These equations describe a dual wheel landing gear with structural damping and consider co-rotating wheels. They represent the system in a fully linearized form. These equations are practically identical to Smiley's equations (11) except for the hysteresis terms for the tire marked by flag marks  (12).

The original 7 degrees of freedom system (3 degrees of translating motion, 3 degrees of rotational motion, 1 degree of tire lateral deformation) has been reduced to a five degree of freedom system assuming constant forward velocity and vertical load and neglecting any degree of freedom in fore and aft (X) and vertical direction (Z). Any coupling between the various modes of tire deformation has also been neglected. The first three equations represent the moment balances of the system. Equations 4 and 5 account for the kinematic relationships. Matrix A is essentially the inertia matrix of the system. Matrix B and C contain the gyroscopic terms of the two wheels as well as the damping terms of the tires and structure. Matrix E represents the elasticity of the system. This set of equations represents a series of five second order differential equations with constant coefficients, if the assumption of constant velocity is applied.

Any stability criterion applied to this set of equations will yield a stability boundary as a function of the various parameters used. It will allow a study of the influence and significance of various parameters upon landing gear stability. It has to be stressed again, however, that these criteria apply rigorously only to the system described by the equations and not necessarily to the actual landing gear.

DEFINITION OF PARAMETERS USED IN TABLES I and II

- A - lateral distance between the two tire center planes
- C_1 - lateral damping coefficient of tire
- C_2 - torsional damping coefficient of tire
- C_3 - time constant of lateral tire motion
- C_4 - drift coefficient of tire
- C_{α} - change in lateral distance of center of pressure in contact area per radian of wheel yaw angle
- C_{λ} - change in lateral distance of center of pressure in contact area per unit lateral deflection
- C_r - change in lateral distance of center of pressure in contact area per radius of wheel tilt
- F - Force
- F_{xc} - lateral ground force of tire
- q - damping factor of shimmy damper
- H_{α} - cornering power of tire
- h - half length of contact area of tire
- $J_{i,j} (i,j = \eta, \gamma, \theta)$ - moments of Inertias of the swiveling part of the landing gear
- I_w - moment of inertia of one tire, wheel, and brake assembly about its axle
- J - decay length of tire
- $K_{i,j} (i,j = \eta, \gamma, \theta)$ - stiffness of landing gear structure
- k_x - fore and aft spring constant of stationary tire
- k_z - vertical spring constant of stationary tire
- k_{α} - torsional spring constant of stationary tire
- k_y - lateral tire force per unit tilt angle
- k_{λ} - lateral spring constant of stationary tire
- L - relaxation length of tire
- M - Moment

- M_{sc} = swiveling moment acting in contact area
- N = cornering power
- r = undeflected radius of tire
- $r_t = r_g$ = torque radius of tire
- V = forward velocity of airplane
- y = horizontal coordinate perpendicular to mean direction of rolling motion
- α_s = static yaw angle of tire
- α_d = drift angle of tire
- γ = roll angle of gear
- ξ = tilt factor of tire
- η = pitch angle of gear (table II)
- η = lateral motion of tire center (table I)
- θ = swiveling angle of gear
- λ_z = lateral deflection of tire contact area
- Δx = fore and aft displacement of tire contact area
- T = constant of proportionate retard of tire
- ω = angular velocity of tire

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